(12) UK Patent Application (19) GB (11) 2 087 025 A

(21) Application No 8035573
(22) Date of filing 5 Nov 1980
(43) Application published 19 May 1982
(51) INT CL[3] F16C 3/28
(52) Domestic classification F2K 4A1 6A
(56) Documents cited GB 762230 GB 233018
(58) Field of search F1N F2K
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(54) Dual-capacity hermetic refrigerant compressor with reversible motor

(57) The invention relates to a dual-capacity hermetic refrigerant compressor of the reciprocating type, the capacity of which can be changed by reversing the crankshaft rotation.

Rotatably mounted on the crankpin (38) of the crankshaft and journaled in the end bearing of each piston rod is an eccentric ring (40) which is coupled to the crankpin in a manner (key 44 in arcuate space 46, 48) resulting in play which permits a limited amount of eccentrically-altering angular relative displacement to occur between the crankpin and the eccentric ring during an initial portion, only, of crankshaft rotation immediately following each direction reversal of the latter, provision being made (elastic key 44 and, optionally, a dashpot effect obtained by providing lubricant passageways 68a) for dampening impacts otherwise resulting when the play in the coupling is taken up.

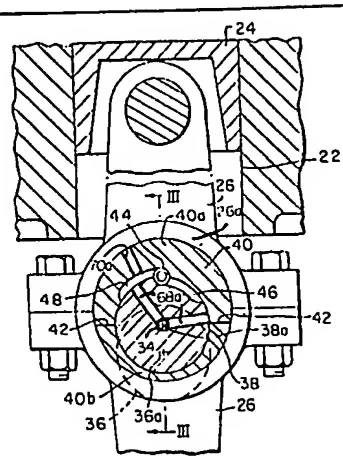

FIG. 2

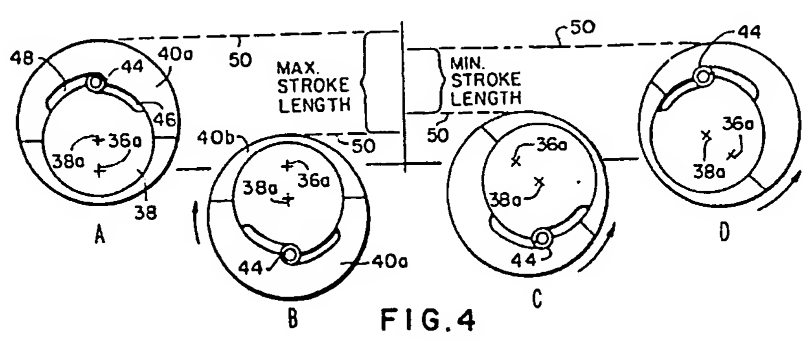

FIG. 4

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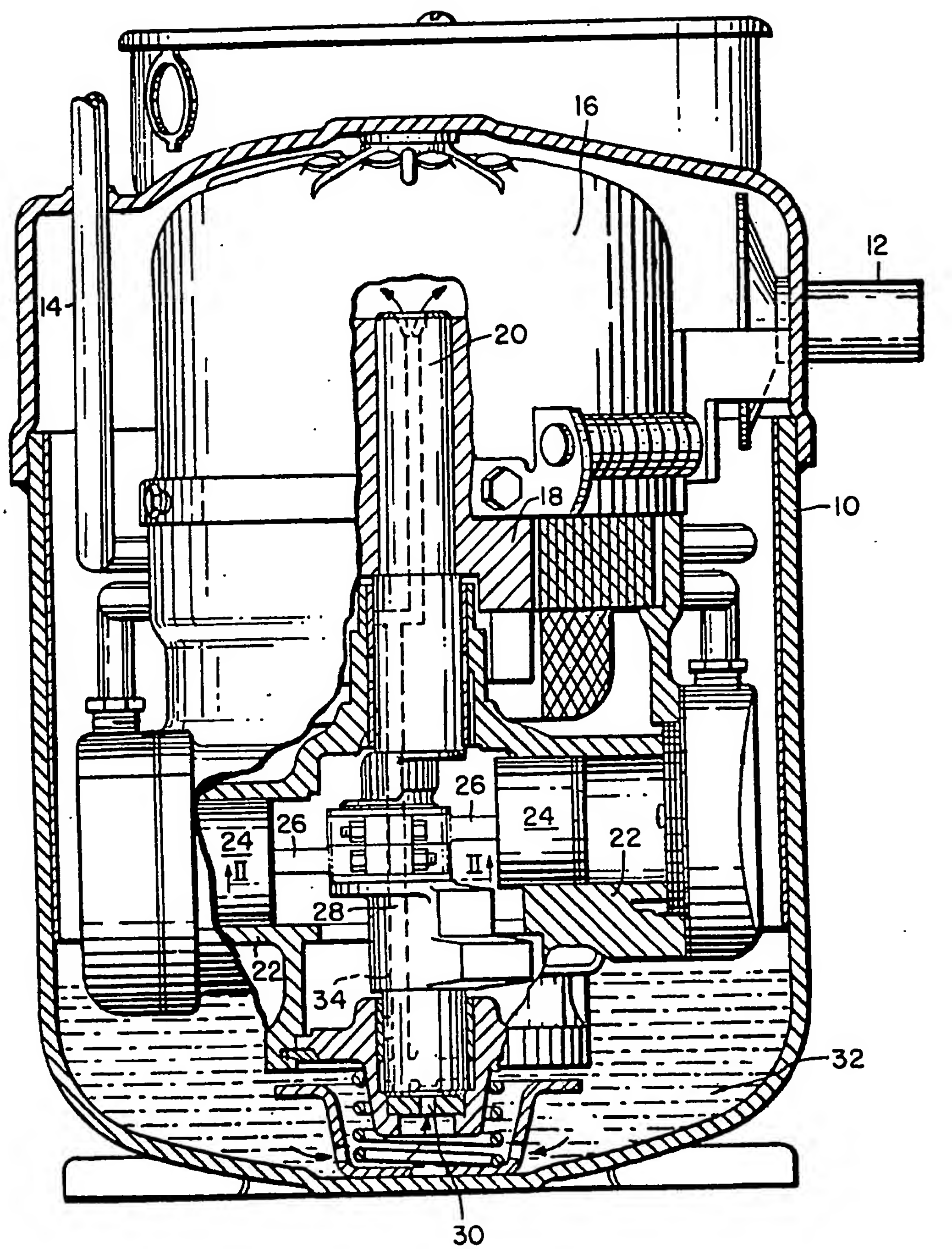

FIG.1

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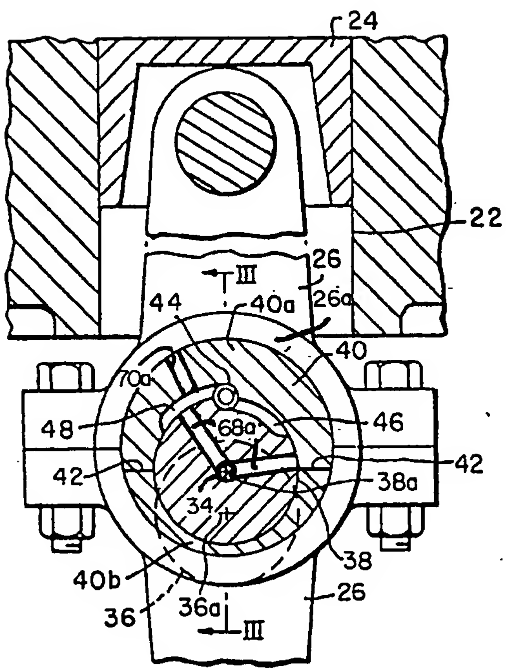


FIG.2

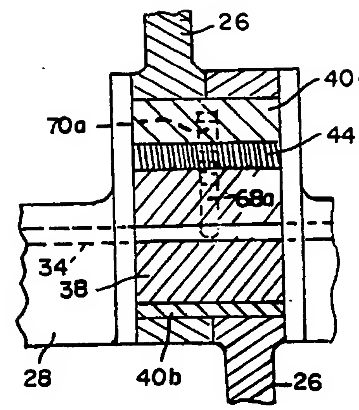


FIG.3

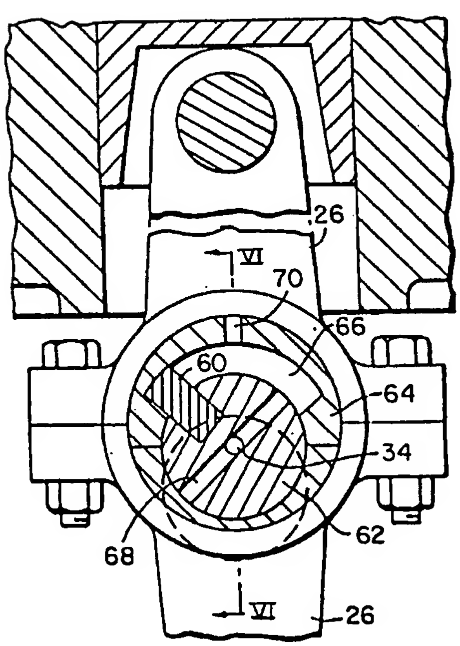


FIG.5

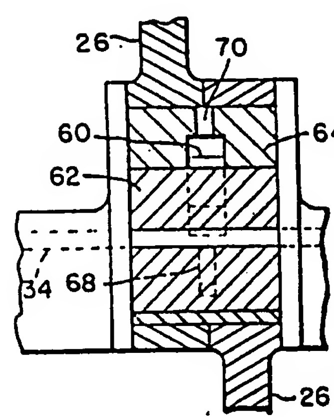
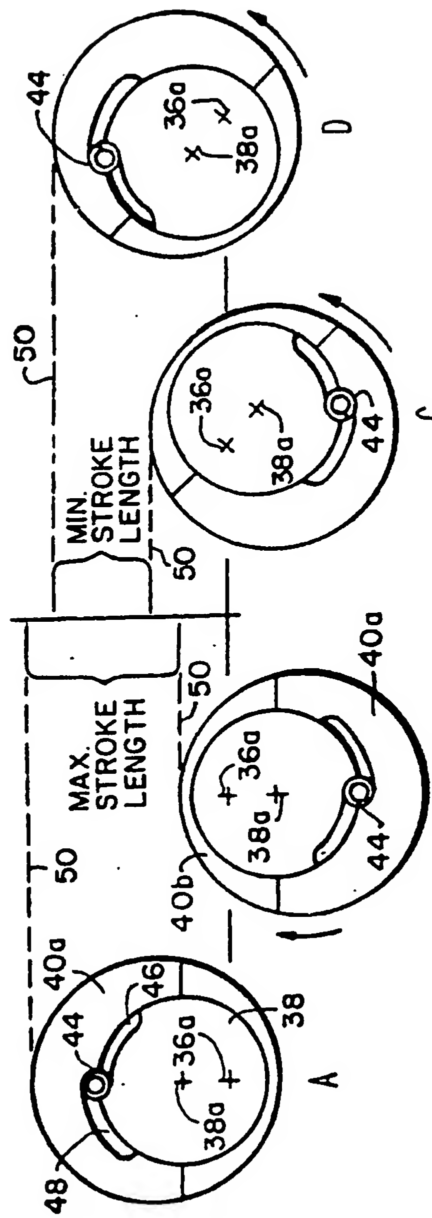


FIG.6



50
MIN.
STROKE
LENGTH
50
MAX.
STROKE
LENGTH
50
50
40a
40b
44
46
48
38
36a
38a
40a
A
B
C
D

FIG. 4

SPECIFICATION

Dual-capacity hermetic refrigerant compressor with reversible motor

The invention relates to a dual-capacity refrigerant compressor for air conditioning units and especially heat pumps.

Studies of heat pump economics have shown that, if a heat pump operating in a heating mode were capable of running efficiently at lower capacity on mild days and at higher capacity on cold days, definite economic advantages would be obtained. The problem is, however, that the compressor capacity of heat pumps operating in a heating mode decreases with falling temperatures because the latter cause the suction gas temperature and density to drop so that less refrigerant is fed to the compressor. Thus, the compressor capacity actually is decreasing when, ideally, it should increase in order to compensate for the lower ambient by supplying more heat. Several ways of handling this problem have been proposed, among them the use of multispeed compressors, of compressors having multiple cylinders which are partially unloadable, and of compressors deliberately oversized to meet all heating needs anticipated (which results in rather poor economics, having regard to the moderate cooling needs experienced in northern regions and to the moderate heating demands to be satisfied on mild days).

From earlier patents it is further known that the output of a pump, or other apparatus employing a reciprocating member, can be changed by changing the eccentricity of the orbiting means driving the connecting rod associated with the reciprocating member. Thus, the U.S. patent specifications Nos. 135,380, 2,592,237 and 3,007,349 teach how such an adjustment of eccentricity can be effected by manually operating a gear or other means, whilst U.S. patent specification No. 3,180,178 provides for eccentricity changes to be brought about by changing the hydraulic pressure in a lubricating system. Of course, it is impracticable to adjust the eccentric means in a hermetically sealed compressor from the outside of the hermetic compressor shell. As for the known arrangement relying upon hydraulic pressure changes, it also has its drawbacks insofar as it calls for hydraulic-fluid actuated means to be mounted on the crankshaft and to rotate with it, and requires the lubrication pressure to be varied by means either of a manually operable pressure regulator or an automatically operated pressure regulator controlled by the load on the compressor, which again would be undesirable with respect to a hermetic system.

Likewise known are arrangements in which variable stroke lengths are obtained through reversals of the operating direction of driving means. Thus U.S. patent specification No. 2,717,518 teaches a direction-sensitive linkage-lengthening arrangement particularly for use with vehicle-windshield wiper mechanisms, whereas in the U.S. patent specification No. 3,482,458 there is disclosed a dual stroke-length mechanism particularly applicable to reciprocating sawing machines in which a pair of links associated with a rotating plate will give different stroke lengths depending upon the rotating direction of the plate. Neither of these arrangements would be suitable for use in a hermetic refrigerant compressor, particularly in view of the considerable difference in magnitude between the forces required in a compressor and those which linkages of these known types could handle.

It is the principal object of the invention to provide a hermetic refrigerant compressor of the reciprocating type with relatively inexpensive yet durable means for automatically adjusting the effective piston-stroke length upon each reversal of crankshaft operation.

The invention accordingly resides in a dual-capacity hermetic refrigerant compressor comprising a rotatable crankshaft having a crankpin which is eccentric with respect to the crankshaft, a reversible motor for driving the crankshaft in either rotary direction, at least one cylinder having a piston therein connected to the crankshaft through a connecting rod having an end bearing encircling the crankpin, and piston-stroke adjusting means for automatically changing the stroke length of the piston upon each directional reversal of crankshaft rotation, said piston-stroke adjusting means comprising an eccentric ring rotatably supported on said crankpin and journaled in said end bearing of the connecting rod, coupling means interposed between the crankpin and the eccentric ring in such manner as to cause the latter, during rotation of the crankshaft, to move together with the crankpin without any angular relative displacement occurring between the crankpin and the ring, said coupling means having a predetermined amount of play enabling a limited eccentricity-altering angular relative displacement to occur between the crankpin and the eccentric ring during an initial portion of crankshaft rotation following each directional reversal thereof, and means effective during said initial portion of crankshaft rotation to dampen impacts otherwise resulting when said predetermined amount of play is taken up.

In a preferred embodiment of the invention, the impact dampening means is an elastic member which serves also as part of the coupling means permitting the limited eccentricity-altering angular relative displacement to occur between the crankpin and the eccentric disc upon each reversal of crankshaft rotation. In another embodiment, impacts are dampened with the aid of a dashpot effect. If desired, such dashpot effect may also be employed with the first-mentioned embodiment to assist the elastic member in dampening impacts.

The preferred embodiment of the invention will now be described, by way of example, in detail with reference to the accompanying drawings, in which:

Fig. 1 is a side elevational and partly

sectional view of a kind of hermetic refrigerant compressor to which the invention is applicable;

Fig. 2 is a detail sectional view, taken along line II—II in Fig. 1, of the preferred embodiment of the invention;

Fig. 3 is a fragmentary sectional view taken along line III—III of Fig. 2;

Fig. 4 is a diagrammatic view illustrating the change in stroke length obtained with the mechanism of Figs. 2 and 3 when the motor drives the crankshaft in one direction and alternatively in the other;

Fig. 5 is a sectional view of another embodiment of the invention; and

Fig. 6 is a fragmentary sectional view taken on line VI—VI of Fig. 5.

The invention is applicable to hermetic refrigerant compressors having either a single cylinder of multiple cylinders, although its best application at present is thought to be with the latter type. The compressor shown in Fig. 1 is of a type similar to the one described in U.S. patent specification No. 3,259,307 to which reference may be had for a more detailed description. Briefly, with reference to Fig. 1, the refrigerant compressor illustrated therein comprises a generally cylindrical, hermetically sealed shell 10 having an inlet 12 through which suction gas refrigerant is admitted to the shell, and one or more discharge tubes 14 through which the compressed gas exits from the shell. The upper part of the shell houses a reversible electric motor 16, the rotor 18 of which is fixed to the upper end of the crankshaft 20.

The compressor is shown as having two cylinders 22 containing pistons 24 which are connected to the crankpin portion of the crankshaft 20 through connecting rods 26.

The lower end portion of the crankshaft 28 includes lubricant inlet means 30 for admitting oil from a sump 32 into a passage 34 extending axially through the crankshaft and carrying oil to the crankshaft bearings, as known.

Referring now to Figs. 2 and 3, the dash-line circle 36 (Fig. 2) indicates the location of the part of the crankshaft 28 which is journaled in the main bearings while the solid-line circle 38 shows the location of the crankpin which is eccentric with respect to said part of the crankshaft, the centerlines of the crankshaft and of the crankpin being indicated at 36a and 38a, respectively. An eccentric ring 40, which derives its eccentricity from its progressively varying wall thickness, is rotatably mounted on the crankpin 38, the ring consisting of two sections 40a and 40b which are held together along the lines 42 by the end bearing 26a of the crankshaft 26, in which end bearing the eccentric ring 40 is journaled. The end bearing 26a likewise is composed of two sections bolted together.

With the arrangement as just described, it will be appreciated that an angular relative displacement of the eccentric ring 40 and the crankpin 38 with respect to one another will result in a change of the total eccentricity affecting the stroke length of the connecting rod 26 and, hence, of the piston 24. As shown in the drawings, the degree of such relative displacement is limited, the means for limiting it comprising, in the embodiment of Figs. 2 and 3, a cylindrical key 44 movably disposed in a space which is defined, between the crankpin 38 and the eccentric ring 40, by a relieved area 46 extending along an arcuate portion of the outer circumference of the crankpin, and a relieved area 48 extending along an arcuate portion of the inner circumference of the eccentric ring. The depth of each of the two relieved areas corresponds substantially to half the diameter of the key 44 or, in other words, the diameter of the latter exceeds the depth of either relieved area 46 or 48 so that any angular relative displacement occurring between the crankpin 38 and the eccentric ring 40 upon initial rotation of the crankshaft in either direction will be terminated due to engagement of the key 44 with one of the opposite end surfaces of the two relieved areas 48, continued crankshaft rotation in the same direction thereafter causing the eccentric ring 40 to be carried along without any further angular relative displacement occurring between it and the crankpin 38. Thus, the key 44 and the end surfaces of the relieved areas 46, 48 in effect form coupling means having a predetermined amount of play which enables a limited, eccentricity-altering angular relative displacement to occur between the crankpin and the eccentric ring during an initial portion of crankshaft rotation following each directional reversal thereof, the coupling means after such limited relative displacement forcing the eccentric ring to move with the crankpin without any further relative displacement occurring therebetween.

Referring now to Fig. 4, there is shown therein the manner in which angular relative displacements of the eccentric ring 40 and the crankpin 38 with respect to each other will change the stroke length of the piston 24. In Fig. 4A, the crankpin and eccentric ring are shown in a top dead-center position, and with the crankpin moving clockwise as indicated by the arrow. Fig. 4B shows the same parts in a bottom dead-center position and still moving clockwise. The dash-line projections 50 toward the center of the drawing indicate the maximum stroke length achieved upon clockwise rotation of the crankshaft together with its crankpin 38.

If the compressor is stopped and is then restarted and driven in the opposite direction by the reversible electric motor 16, the crankpin 38 will first turn within the eccentric ring 40 counterclockwise until the key 44 reaches and engages the opposite end surface or shoulder of the relieved area 48 of the ring 40, as shown in Fig. 4C, and, upon further counterclockwise rotation, drives the eccentric ring 40 along to move together with the crankpin 38, as indicated in Fig. 4D. It can be seen from Fig. 4 that this directional reversal of crankshaft rotation and the consequential angular relative displacement between the crankpin and the eccentric ring have

resulted in a reduction of the stroke length. A subsequent reversal of the crankshaft rotation will increase the stroke length again, as shown in the left-hand portion of Fig. 4.

In the embodiment illustrated in Figs. 5 and 6, a key 60 is fixed to the crankpin 62 and projects radially therefrom into a recess 66 which, extending through a predetermined angle, is formed in the inner circumference of the eccentric ring 64 and, together with a portion of the outer circumference of the crankpin 62, defines an arcuate space in which the projecting portion of the key 60 is movable. It is conceivable that the arrangement could be reversed in the sense of providing a recess similar to recess 66 in the outer circumference of the crankpin 62, and affixing a key similar to the key 60 to the eccentric ring 64 so that a portion of the key would extend radially from the inner circumference of the ring and into the recess of the crankpin.

In order to minimize wear and tear resulting from impacts occurring when the play in the coupling means between the crankpin and the eccentric ring is taken up, provision is made for dampening such impacts. In the embodiment according to Figs. 5 and 6, the means for dampening impacts comprises passageways 68 formed in the crankpin 62 and extending from the previously mentioned lubricant supply passage 34 radially in opposite directions to the outer peripheral surface of the crankpin. Another passageway 70 formed in the eccentric ring 64 is provided to permit lubricant to flow from the arcuate space 66 to the bearing surfaces between the eccentric ring and the end bearing of the connecting rod 26. The passageways 68 and 70 are arranged in such manner as to place the arcuate space 66 in fluid-flow communication with the lubricant supply and with said bearing surfaces throughout rotation of the crankshaft, yet to impede lubricant flow from the arcuate space 66 during an initial portion of crankshaft rotation immediately following a directional reversal of the latter, thereby providing an impact dampening dashpot effect in the following manner.

Supposing the compressor was stopped, with the key 60 positioned as shown in Fig. 5, and is now restarted with the shaft rotating in the clockwise direction. As the crankpin begins to rotate clockwise within the still stationary eccentric ring, the key 60 will push oil in the arcuate space 66 ahead of it and out through the passageway 70 as well as into the oil passage 68. When the key has moved past the passageway 70 and the passageway in the crankpin has moved out of registry with the space 66, any oil remaining in the latter ahead of the key 60 can escape only through normal clearances in the assembly, thus encountering a high flow resistance which produces the above-mentioned dashpot effect. When all of the oil ahead of the key 60 has been slowly forced from the space 66 through these clearances, the key will have reached the opposite end of the space 66 and have gently engaged the adjacent end surface of the recess in the eccentric ring. At the same time, the opposite end of the passageway 68 will have come into registry with the space 66, thus reestablishing normal lubricant flow from the passage 34 into the space 66. When the compressor is stopped again and restarted in the opposite direction, the same sequence will occur in reverse.

In the embodiment illustrated in Figs. 2 and 3, impact dampening is achieved by means of the key 44 which is so constructed as to have a sufficient degree of elasticity to absorb impact forces. Thus, the key 44, as shown, is in the form of a spiral spring which will effectively cushion impacts due to the play in the coupling means between the crankpin 38 and the eccentric ring 40 being taken up. It is conceivable that a key consisting of a cylindrical member made of a suitable elastomer which is sufficiently heat resistant, strong and elastic might be employed instead of a spiral spring.

If desired, the crankpin 38 and the eccentric ring 40 of the embodiment illustrated in Figs. 2 and 3 may be provided, as indicated in the latter, with passageways 68a and 70a, respectively, enabling a dashpot effect to be obtained in a similar manner as described above. Such dashpot effect, whilst probably not required in most instances in which the embodiment utilizing the elastic key 44 is employed, would assist the latter in dampening impacts.

As mentioned hereinbefore, the invention is applicable to both single-cylinder and multiple-cylinder types of compressors, the compressor to which it is shown applied herein having two cylinders. Referring in this connection to Figs. 3 and 6, it will be seen therefrom that the arrangement embodying the invention is readily adaptable for use with multiple cylinders by using a crankpin and eccentric ring having axial dimensions sufficient to accommodate the end bearings of all connecting rods side-by-side in axial alignment with each other. With this arrangement, the single ring 40 or 64 serves to adjust the stroke length of all connecting rods 26 simultaneously.

It will be appreciated that the degree of stroke reduction may vary from case to case, depending upon the degree of eccentricity of the eccentric ring selected for use. An example would be a stroke reduction of, say, about 30%. With a high-capacity stroke length of unity, and with a clearance ratio of, say, 5%, this 30% stroke reduction, when effected by reversing the operating direction of the compressor, will result in a new clearance ratio of 28.6% for the reduced stroke length. It will be appreciated and is apparent from Fig. 4 that with the arrangement as shown, the reduction of the stroke length occurs both at the top dead-center position and the bottom dead-center position.

A control arrangement particularly suitable for use in controlling and reversing a hermetic refrigerant compressor such as described herein, especially when employed with heat pumps, is

disclosed and claimed in Applicant's copending application No. (WE-47, 635-I).

CLAIMS

1. A dual-capacity hermetic refrigerant compressor comprising a rotatable crankshaft having a crankpin which is eccentric with respect to the crankshaft, a reversible motor for driving the crankshaft in either rotary direction, at least one cylinder having a piston therein connected to the crankshaft through a connecting rod having an end bearing encircling the crankpin, and piston-stroke adjusting means for automatically changing the stroke length of the piston upon each directional reversal of crankshaft rotation, said piston-stroke adjusting means comprising an eccentric ring rotatably supported on said crankpin and journaled in said end bearing of the connecting rod, coupling means interposed between the crankpin and the eccentric ring in such manner as to cause the latter, during rotation of the crankshaft, to move together with the crankpin without any angular relative displacement occurring between the crankpin and the ring, said coupling means having a predetermined amount of play enabling a limited eccentricity-altering angular relative displacement to occur between the crankpin and the eccentric ring during an initial portion of crankshaft rotation following each directional reversal thereof, and means effective during said initial portion of crankshaft rotation to dampen impacts otherwise resulting when said predetermined amount of play is taken up.

2. A dual-capacity hermetic refrigerant compressor according to claim 1, wherein said crankpin includes a relieved area along an arcuate portion of its outer circumference, and said eccentric ring includes a similar relieved area along an arcuate portion of its inner circumference, said relieved areas cooperating to define an arcuate space in which is disposed a cylindrical key which has its longitudinal axis substantially parallel to the longitudinal axis of the crankpin, and has a diameter exceeding the depth of either one of said relieved areas, said cylindrical key being movable in said space between end surfaces of said relieved areas and, together with said end surfaces, forming said coupling means.

3. A dual-capacity hermetic refrigerant compressor according to claim 3, wherein said cylindrical key is elastic and forms the impact dampening means.

4. A dual-capacity hermetic refrigerant compressor according to claim 2 or 3, wherein said cylindrical key is a spiral spring.

5. A dual-capacity hermetic refrigerant compressor according to claim 1, wherein one of the members constituting said crankpin and eccentric ring has a recess formed in an arcuate portion of the outer or inner, respectively, circumference thereof which recess, together with the inner or outer, respectively, circumference of the other member, defines an arcuate space, said other member carrying a key having a portion thereof projecting into said arcuate space and being movable therein between opposite end surfaces of said recess, said key and said end surfaces of the recess cooperating to form said coupling means.

6. A dual-capacity hermetic refrigerant compressor according to claim 2, 3, 4 or 5, wherein said crankpin has passageways formed and arranged therein in such manner as to maintain said arcuate space in fluid-flow communication with a lubricant supply during rotation of the crankshaft and to impede lubricant flow from said arcuate space and thereby produce an impact-dampening dashpot effect in the latter part of said initial portion of crankshaft rotation following each directional reversal of the latter.

7. A dual-capacity hermetic refrigerant compressor according to any of the preceding claims, including at least one additional cylinder having a piston therein connected to said crankshaft through a connecting rod which has an end bearing, wherein said crankpin and said eccentric ring thereon have an axial dimension sufficient to accommodate the end bearings of all connecting rods with said end bearings disposed side-by-side in axial alignment with each other and embracing the eccentric ring.

8. A dual-capacity hermetic refrigerant compressor substantially as hereinbefore described with reference to, and as illustrated in, the accompanying drawings.

Printed for Her Majesty's Stationery Office by the Courier Press, Leamington Spa, 1982. Published by the Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.